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CRANK CASE SCAVENGING OF TWO-STROKE-CYCLE ENGINES

By Hans List

From Zeitschrift des Vereines deutscher Ingenieure
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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS.

TECHNICAL MEMORANDUM NO. 516.

CRANK CASE SCAVENGING OF TWO-STROKE-CYCLE ENGINES.*

By Hans List.**

Small two-stroke-cycle Diesel engines with crank case scavenging must function at very low mean piston pressures for the combustion to be complete. This is due to the poor scavenging of these engines and the consequent small amount of air available for the combustion. The quantity of scavenging air is limited by the fact that the inner side of the piston operates as a pump plunger and cannot therefore greatly exceed the piston displacement. This quantity of air is insufficient for the complete removal of the exhaust gases from the cylinder, because large quantities of the scavenging air are lost through the exhaust ports during and immediately after the scavenging. These losses must be reduced, in order to effect any considerable improvement in the scavenging. The efficiency of the crank case scavenging pump is high, especially when the dynamic effect of the exhaust-gas column is utilized. It is generally between 95 and 100%, even when no special provisions are made, as the writer was able to demonstrate by accurate air measurements in testing a two-stroke-cycle Diesel engine.

*"Kurbelkastenspülung für Zweitaktmotoren" from Zeitschrift des Vereines deutscher Ingenieure, February 16, 1929, pp. 225-229.

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In order to calculate the attainable mean piston pressure, we must know the quantity of air available for the combustion. It can be calculated, when the air consumption, the efficiency of the scavenging and the length of the exhaust port are known. By "air consumption" is meant the ratio of the scavenging air to the piston displacement, while the efficiency of the scavenging is the ratio of the air remaining in the cylinder after the scavenging to the total amount of air taken in.

In the case of scavenging with nearly constant pressure, the air consumption can be determined from this pressure, the cylinder and port dimensions and the revolution speed, if we know the coefficient of flow for the system comprising the inlet passage, cylinder and outlet passage. This empirical factor can be estimated, in the case of projected engines, from the results obtained with similar engines, while, in the case of engines actually built, it can be determined by air measurements. In engines with crank case scavenging, the scavenging pressure diminishes, however, during the scavenging, from the maximum value nearly to zero. The relations here are somewhat more difficult than in the case of constant scavenging pressure, but can likewise be determined theoretically with a knowledge of the coefficient of flow.

If σ_e is the ratio of the length of the scavenging ports to the stroke s , the piston will open the ports when it is at a distance of $s \sigma_e$ from the lower dead center. Since the

piston reverses its direction during the scavenging, and since the piston speed is small in comparison with the velocity of the air and exhaust gases, the piston motion has no great effect on the scavenging. We may therefore disregard it and assume that the scavenging air flows from a container of constant capacity coinciding with the mean capacity of the crank case during the scavenging. Since this differs but little, however, from the capacity V_r of the crank case at the inner piston dead center and since the uncertain bases of the calculation entirely justify this simplification, we can consider the piston as stationary at the inner dead center during the scavenging. Let V_h denote the piston displacement. Then, with $V_r/V_h = x$, the compression ratio becomes

$$\frac{V_r}{V_r + V_h} = \frac{x}{x + 1}.$$

In adiabatic compression the final compression pressure is

$$p_r = p_a \eta_v \left(\frac{x + 1}{x} \right)^k,$$

in which $p_a \eta_v$ is the pressure at the beginning of the compression, p_a being the external pressure and η_v the valve resistance. For the present we will consider η_v as having a value of 1.

If the capacity of the crank case increases during the scavenging, the scavenging pressure decreases. For the flow from a container of constant capacity and given initial pres-

sure, we have, according to Schüle (Technische Thermodynamik, second edition, Volume II, p.524 ff.), the following relation between the time integral of the area $\int f \, d \, t$ and the pressure p_i' in the container.

$$\int f \, d \, t = \frac{1}{k} \frac{V}{\varphi} \frac{1}{\sqrt{p_r v_r}} \left(\frac{p_r}{p_a} \right)^{\frac{1}{2} - \frac{1}{2k}} p_i \int_{p_r}^{p_i} \frac{1}{\left(\frac{p_a}{p_i'} \right)^{\frac{1}{2k} + \frac{1}{2}} a} d \left(\frac{p_a}{p_i'} \right)$$

It is here assumed that the expansion takes place adiabatically. This does not agree exactly with the reality, but the discrepancy only slightly affects the result, since the resulting error is much smaller than the approximation with which (for example) φ can be estimated. In the formula, V denotes the capacity of the container, p_r its initial pressure, p_a the external pressure, v_r and v_a the corresponding specific volumes, φ the coefficient of flow, p_i the pressure at the closing of the ports, k the exponent of the adiabatic line and

$$a = \sqrt{2g \frac{k}{k-1} \left[\left(\frac{p_a}{p_i'} \right)^{\frac{1}{k}} - \left(\frac{p_a}{p_i'} \right)^{\frac{k+1}{k}} \right]}$$

If the expression under the integral sign is plotted as a function of p_a/p_i' , then, according to Schüle, $z = f(p_a/p_i')$ the integral curve and

$$\int f \, d \, t = \frac{1}{k} \frac{V}{\varphi} \frac{1}{\sqrt{p_r v_r}} \left(\frac{p_r}{p_a} \right)^{\frac{1}{2} - \frac{1}{2k}} (z_i - z_r)$$

in which z_i and z_r correspond, respectively, to the abscissas p_a/p_i and p_a/p_r . The weight of air removed from the crank case, while the pressure is dropping from p_r to p_i , is

$$G = \frac{V_h x}{v_a} \left(\frac{v_a}{v_r} - \frac{v_a}{v_i} \right)$$

Now

$$\frac{v_a}{v_r} = \left(\frac{p_r}{p_a} \right)^{\frac{1}{k}}, \quad \frac{v_a}{v_i} = \left(\frac{p_i}{p_a} \right)^{\frac{1}{k}}$$

and hence

$$G = \frac{V_h x}{v_a} \left[\left(\frac{p_r}{p_a} \right)^{\frac{1}{k}} - \left(\frac{p_i}{p_a} \right)^{\frac{1}{k}} \right]$$

If the air is taken from the outside atmosphere, its quantity is

$$V_a = V_h x \left[\left(\frac{p_r}{p_a} \right)^{\frac{1}{k}} - \left(\frac{p_i}{p_a} \right)^{\frac{1}{k}} \right].$$

If we put

$$V_a = \lambda V_h,$$

in which λ is the air consumption in the scavenging at V_h , then

$$\frac{p_i}{p_a} = \left(\frac{x + 1 - \lambda}{x} \right)^k, \quad \frac{p_r}{p_a} = \left(\frac{x + 1}{x} \right)^k.$$

From this, when x and λ are given, z_i and z_a can be determined with the aid of the integral curve.

For $\int f d t$ we find, if ψD is the width of the ports, σ' the momentarily free length of the slot and r/l the connecting rod ratio, a second expression

$$\int f d t = \psi D s \int \sigma' d t$$

and with

$$d t = \frac{30}{\pi n} d a$$

and

$$\int \sigma' d a = \left(1 + \frac{r}{2l}\right) \frac{\sigma_e^{3/2}}{370} \quad (\text{according to Föppl})$$

$$\int f d t = \psi D s \frac{30}{\pi n} \left(1 + \frac{r}{2l}\right) \frac{\sigma_e^{3/2}}{370}.$$

By making the two expressions equal, we obtain

$$x (z_i - z_r) = \frac{4.3 \varphi \psi \left(1 + \frac{r}{2l}\right)}{D n} \sigma_e^{3/2} = 4.3 A \sigma_e^{3/2} = U$$

The left side of the equation is a function of x and λ , while the right side depends on empirical and constructional values. σ_e must be expressed in per cent.

In Figure 1, λ is represented as a function of x and U , while U is brought into relation with A and σ_e . We begin with A in order to find λ for given conditions. We go toward the left, until we intersect the corresponding line σ_e , then upward to the intersection with the line of the fixed value of x , and can then read at the left the value of λ . Then, after estimating the coefficient of flow φ , we can calculate the air consumption.

The losses of scavenging air are due to the fact that it escapes with the exhaust gases and that a portion of the cylinder content is displaced by the piston. This displacement generally occurs after the completion of the scavenging. This is especially true of scavenging with falling pressure, but for the present deductions it is sufficiently accurate even for scaveng-

ing at constant pressure. We can therefore assume that the two processes are practically separate in point of time and consider their losses separately.

The air lost during the scavenging cannot be calculated, because the flow and mixing conditions in the cylinder cannot yet be mathematically determined. This part of the scavenging process can be elucidated therefore only by experimentation, either by investigating the flow in the cylinder or by analysis of the cylinder content before and after the scavenging, in order to determine its efficacy. The investigation of the flow furnishes information as to what points are not swept by the scavenging air, but, since the mixing conditions are not included, furnishes only unsatisfactory bases for determining the scavenging effect, which is the only important consideration.

A p p a r a t u s

Moreover, the determination of the scavenging efficiency by analysis is troublesome and inaccurate, especially when it is desired to investigate the effect of various shapes and sizes of cylinders. I have therefore designed an apparatus with which systematic investigations can be made of the scavenging in small cylinders. Theoretically, the scavenging efficiency depends on the arrangement of the scavenging and exhaust ports, the shape of the piston head and cylinder cover, the stroke/bore ratio, the scavenging pressure, the cylinder dimensions, the law of

opening of the scavenging cross-section areas and the quantity of scavenging air. We must be able to vary all these quantities, in order to make a thorough investigation of the scavenging.

The cylinder model a (Fig. 2) is filled with carbon dioxide and then scavenged with a given quantity of air, the carbon dioxide remaining in the cylinder being measured by absorbing it in a solution of potassium hydroxide. The cylinder model is so divided that the arrangement of the ports, the stroke/bore ratio and the shape of the piston head and cylinder cover can be changed by exchanging individual parts. The inflowing air is regulated by a valve b, which opens under the action of a falling weight c, according to the same law, after the piston uncovers the ports. The scavenging-air container d has the same capacity as the dead space in the crank case scavenging pump. It is filled to the final compression pressure. By varying its content and pressure, the scavenging conditions can be investigated for various dead spaces in the crank case scavenging pump. The capacity of the container can be diminished by filling it partially with water. The opening time and time integral of the area of the scavenging valve can be regulated by the falling distance of the weight serving to open the valve and by the tension of the valve spring.

The functioning of the model differs from the scavenging in the engine, in that the inlet passage e is scavenged simultaneously and the capacity of the outlet passage f, up to the

upper edge of the exhaust port, is measured with it. The experimental results are freed from these unavoidable errors by a corrective process. In the representation of the results, the air consumption is plotted on the abscissas and the scavenging efficiency on the ordinates. The scavenging efficiency η_1 is here the ratio of the quantity of air in the cylinder immediately after the scavenging to the piston displacement.

Five arrangements of the scavenging ports were tried, four of which are shown in Figures 3-10. The last arrangement (IV) corresponds to that of a Junkers engine. For structural reasons it does not come under consideration as regards the crank case scavenging, but is introduced for comparison.

Loop scavenging (Figs. 3-8).-- Figure 11 shows the volumetric and scavenging efficiencies for scavenging and exhaust ports situated one above the other, as represented in Figures 3 and 4 (I). The experiments yielded approximately identical values for the stroke/bore ratios 1.2, 1.48, 1.76, and 1.96. Also in Figure 12, which corresponds to Figures 5-6 (II) for two scavenging ports in the middle and two lateral exhaust ports, the stroke/bore ratios of 1.1 and 1.8 practically coincided. On the other hand, in Figure 13, for two exhaust ports in the middle and two scavenging ports on the sides, Figures 7-8 (III), the volumetric efficiency diminished somewhat with increasing stroke/bore ratio. It is recognized, however, that all three arrangements are well suited to large stroke-bore ratios.

Through scavenging.-- Figure 14 shows the results of experiments with a flat piston head (Figs. 9-10 (IV)). The volumetric efficiency decreases greatly with increasing stroke/bore ratio and is always considerably smaller than with loop scavenging. The slightly conical piston head and a piston head hollowed out for deflecting the scavenging air gave similar results.

Scavenging with Junkers double-piston engine.-- Figure 15 shows the remarkably high volumetric efficiency of the scavenging, which also follows quite a different course from what it does in the other arrangements.

Analysis of the Results

The effect of the shape of the cylinder cover on the volumetric efficiency was investigated with the retention of the other quantities and relations. With loop scavenging and through scavenging, as well as with large and small stroke/bore ratios, the discrepancies were very small for differently shaped cylinder covers. The volumetric efficiency is therefore practically independent of the shape of the cylinder cover for all the scavenging arrangements. This was also found to hold good for the law of opening and for the initial scavenging pressure.* Here also the volumetric efficiencies are practically equal under otherwise like conditions. In all the arrangements the

*This applies only to crank-case scavenging. At nearly constant scavenging pressure, recent investigations show the dependence of the volumetric efficiency on the scavenging pressure.

course of the volumetric efficiency was sensitive to changes in the position of the incoming air stream with respect to the outgoing gas stream. In some arrangements of the ports (e.g., according to Figs. 7-8), the volumetric efficiency could be much improved only by reducing the height of the exhaust ports. With the same means, on the other hand, this result was less attainable by uniflow or through scavenging.

From its independence of the scavenging pressure and hence of the inflow velocity, it may be concluded that the volumetric efficiency, within a limited range, is also independent of Reynolds Number and therefore of the dimensions. Strictly speaking, the values hold good only for the cylinders tested. They render it possible, however, to determine the nature of the volumetric efficiency's independence of the air consumption and to appraise the various arrangements of the ports. It is intended to verify the results on a running engine by means of a special experimental device for removing and analyzing the entire contents of the cylinder.

From the efficiency curves of the loop and through scavenging, it follows that the scavenging efficiency increases nearly proportionally for all port arrangements and stroke-bore ratios with the air consumption inside the limits of 60-110%, which alone come under consideration for the crank-case scavenging. We can therefore write $\eta_s = a - C \lambda$ and $\eta_l' = a \lambda - C \lambda^2$ and can follow the curve through two experiments.

The scavenging efficiency is dependent in practice only on the guidance of the scavenging air in the lower part of the cylinder. In many arrangements the course of the curve is affected by the stroke/bore ratio, but never by the initial pressure. No general rules can now be given aside from the ones already known, such, for example, as the rule that there must be no short-circuit flow. The best way is always to investigate thoroughly any contemplated arrangement, as in the matter of airplane wing profiles, and determine the most favorable conditions. The tests should also be continued at a nearly constant scavenging pressure.

After finding the volumetric efficiency curve for a given course of the scavenging air, we can calculate approximately the quantity of air available for the combustion. After the scavenging, more air remains in the cylinder than corresponds to the volumetric efficiency, because the exhaust gases also contain air, due to the combustion with an excess of air. The proportion of air in the exhaust gases is

$$\frac{L - 12}{L + 0.7}$$

when $L \text{ m}^3$ of air is available for 1 kilogram of fuel and the theoretical quantity of air is $12 \text{ m}^3/\text{kg}$. The total amount of ^{air} remaining in the cylinder after the scavenging is then

$$[(1 - \eta_v') \frac{L - 12}{L + 0.7} + \eta_v'] V_h$$

in which τ is a temperature coefficient for reducing the content of the exhaust gases to the condition of the scavenging air. The piston forces about $1/\sigma_a$ of this air through the exhaust ports. Hence the total volumetric efficiency is

$$\eta_l = [(1 - \eta_l') \frac{L - 12}{L + 0.7} \tau + \eta_l'] (1 - \sigma_a).$$

The mean piston pressure attainable with $L = 18 \text{ m}^3/\text{kg}$ can be estimated at

$$p_e = 5.4 [0.21 + 0.79 \eta_l'] (1 - \sigma_a)$$

for small two-stroke Diesel engines with the values of the other quantities customary for these engines. It is assumed, however, that p_e increases with the quantity of air, and the calorific value of the mixture is not considered.

The formula for the volumetric efficiency is not quite accurate because, for lack of empirical values, the effect of heating the scavenging air is not taken into account. The error cannot, however, be very great and is about the same for all kinds of scavenging.

If we plot p_e against σ_e , we obtain a curve with a high maximum value, which corresponds to the most favorable length of the scavenging ports. A two-stroke engine with crank case scavenging might have, e.g., a bore of 120 mm (4.72 in.), a stroke of 180 mm (7.09 in.), a revolution speed of 550 R.P.M., and a connecting-rod ratio of 1/4. The dead space of the crank case scavenging pump is 500%. The dependence of the mean piston

pressure on the length of the scavenging ports was tested for the port arrangements III and IV. In both cases the width of the ports was 69% of the cylinder diameter; the length of the exhaust ports, 20%; and the length of the scavenging ports, 10%. For other lengths of the scavenging ports, the lengths of the exhaust ports were such that the time integral of the area for the pressure-drop period remained constant. φ was estimated at 0.65.

In Figure 16 the air consumption, volumetric efficiency of the scavenging and the mean piston pressure for the port arrangements III and IV are plotted against σ_e . It is seen that, in both cases, the maximum mean pressure lies at about 10% of the length of the scavenging ports and that the curve is very flat. The diagram shows that the length of the scavenging ports, over a large range, has but little effect on the mean piston pressure. The superiority of the port arrangement III is demonstrated in the increase of the mean pressure from 2.7 to 3.2.

Translation by Dwight M. Miner,
National Advisory Committee
for Aeronautics.

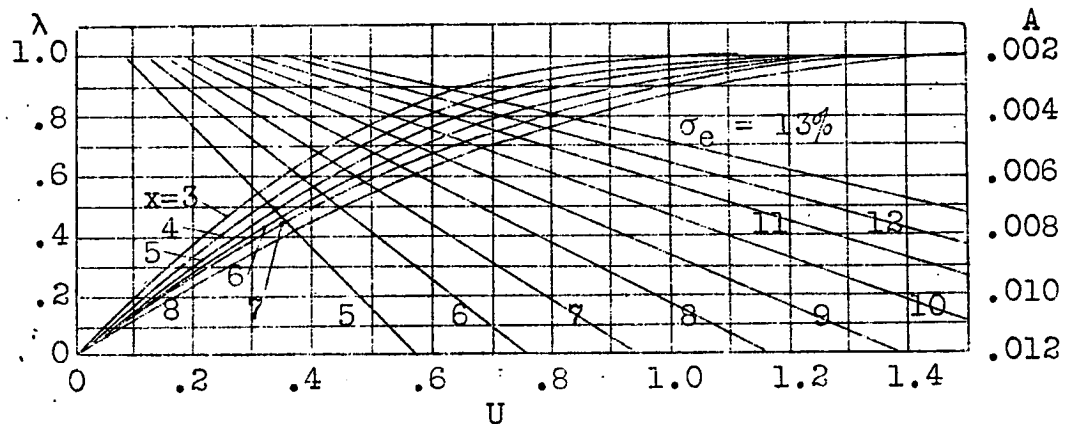


Fig.1 Air consumption in crankcase scavenging.

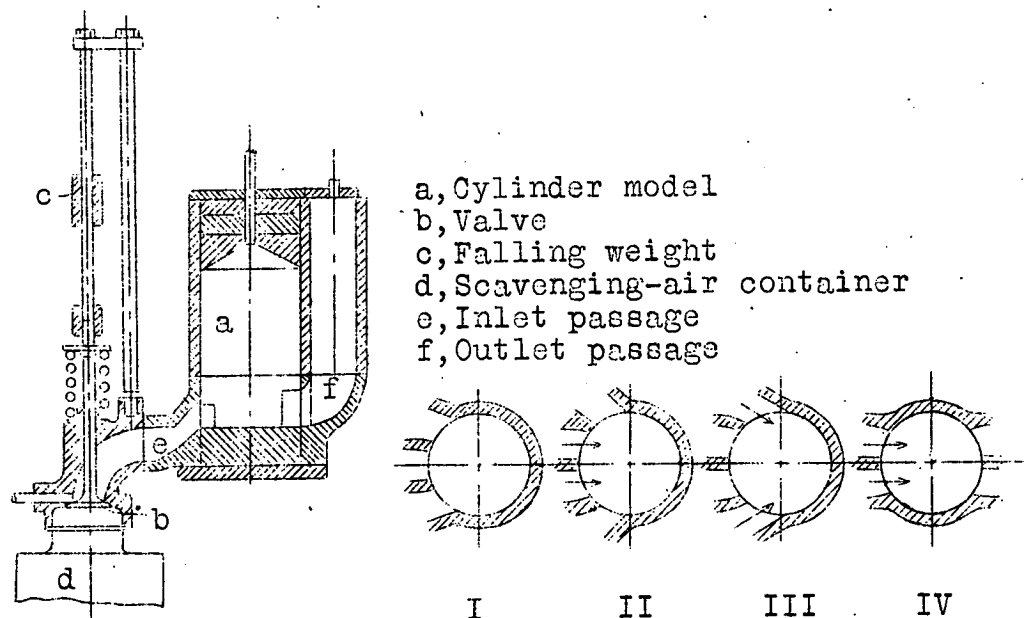


Fig.2 Cylinder model and shut-off valve for scavenging investigations.

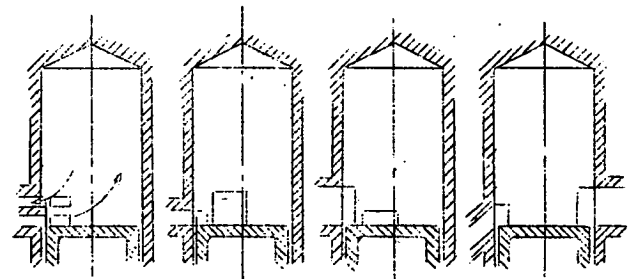
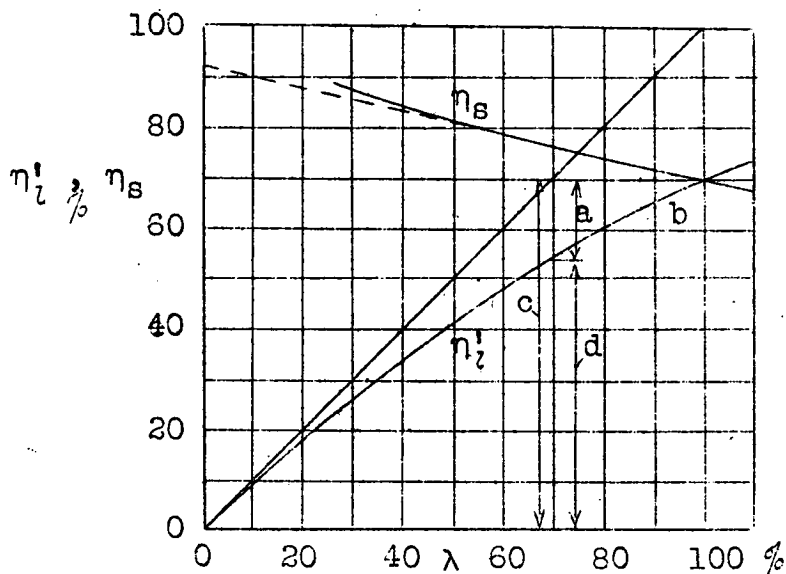


Fig.3 to 10 Tested port arrangements I to IV.



a, Lost air

d, Scavenging air left in cylinder

b, $\frac{s}{D} = 1.3$ to 1.95

c, Total scavenging air

Fig.11 Slot arrangement.

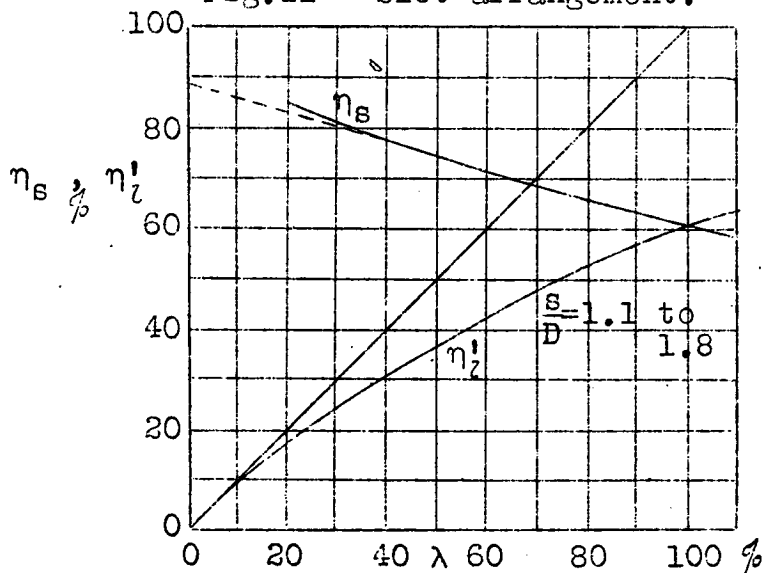


Fig.12 Port arrangement II.

Figs.11 & 12 Scavenging efficiency η_s and volumetric efficiency η'_v of the scavenging for different lengths and arrangements of the scavenging ports.

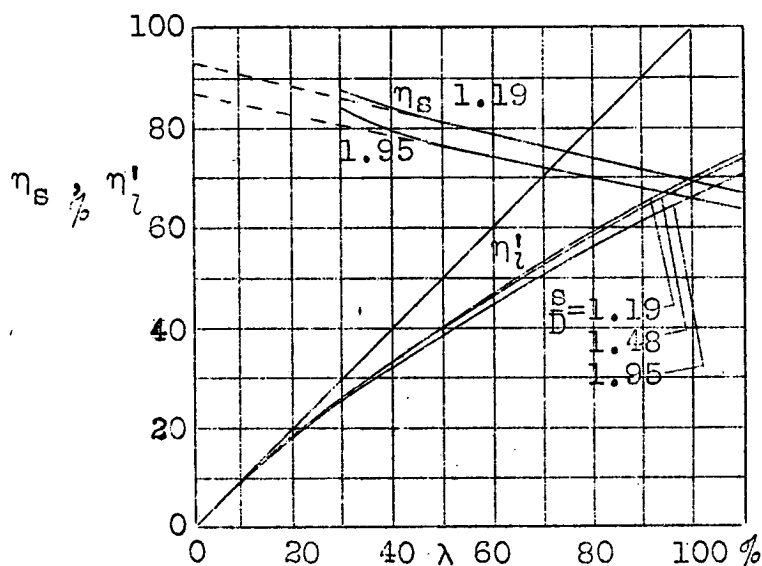


Fig.13 Port arrangement III.

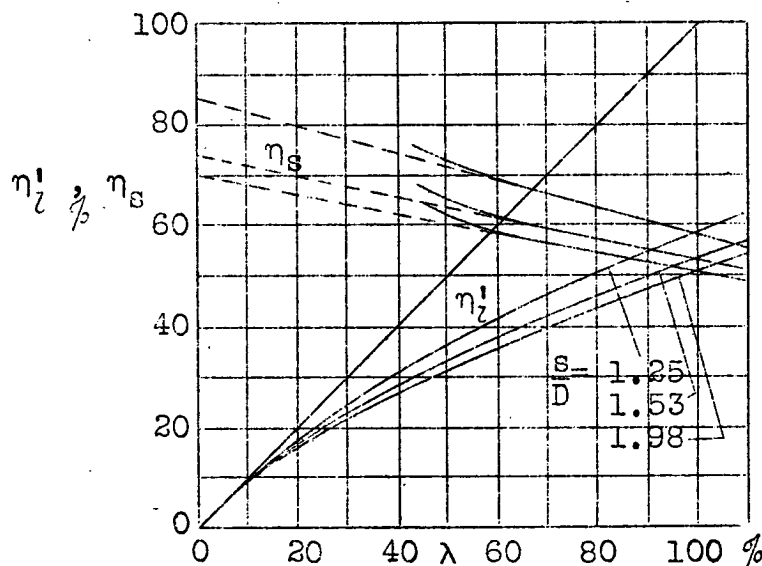


Fig.14 Port arrangement IV.

Figs.13 & 14 Scavenging efficiency η_s and volumetric efficiency η'_l of the scavenging for different lengths and arrangements of the scavenging ports.

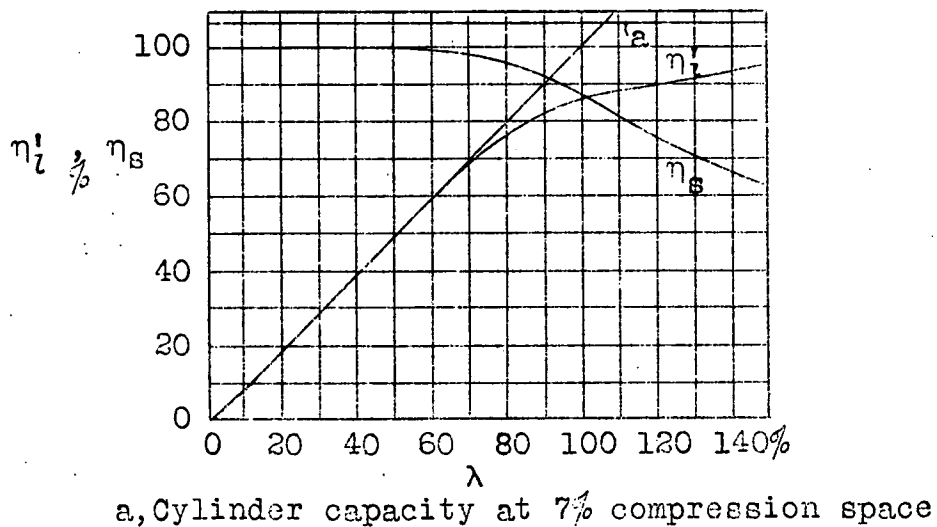


Fig.15 Port arrangement V, (Junkers). Scavenging efficiency η_s and volumetric efficiency η_v of the scavenging for different lengths and arrangements of the scavenging ports.

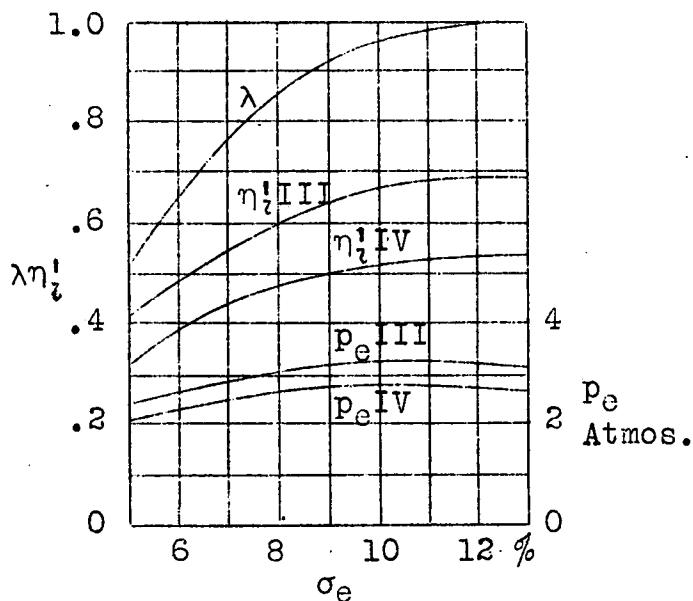


Fig.16 Air consumption, volumetric efficiency and mean piston pressure plotted against the length of the scavenging ports. III and IV indicate the port arrangement.